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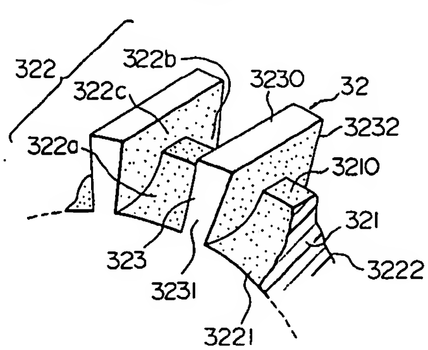
(54) **Fuel pump.**

(57) A Westco type fuel pump includes an impeller (32) which has a plurality of vane grooves (322) and a plurality of vane plates (323) provided alternately along its outer periphery. Each vane groove (322) is constituted by groove portions (322a, 322b) formed in both sides of the impeller (32), respectively, with a partition wall (321) provided between the groove portions (322a, 322b). The partition wall has an outer peripheral surface (3210) located radially inside an outer peripheral surfaces (3230) of each vane plate (323) and has a predetermined thickness in an axial direction of the impeller. As the impeller (32) rotates, two vortex flows of fuel are generated along bottom surfaces (3221, 3222) of the groove portions (322a, 322b) and then smoothly merge together at a position outside the outer peripheral surface (3210) of the partition wall, thereby reducing a flow dead zone (96) in a pump flow passage (33). When the impeller (32) is molded by using molds, deformation of the molded impeller is prevented due to the thickness of the outer peripheral surface (3210). Of the surfaces of the impeller (32), therefore, the surfaces of each vane groove remain as they are after the molding, while both sides of the impeller (32) and the outer peripheral surface (3230) of the vane plates (323) are ground. Thus, the impeller (32) able to surely achieve a high level of pump performance can be easily provided by resin molding.

FIG. 1



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BACKGROUND OF THE INVENTION

The present invention relates to a fuel pump for feeding fuel to an internal combustion engine or the like. This pump is used, for example, to supply fuel under pressure to a fuel injection system in an automobile or the like.

An automobile or the like having an engine equipped with a fuel injection system of electronic control type employs a motor-operated fuel injection pump as one part of a device for injecting fuel to the engine. The fuel pump is dipped in a liquid fuel contained in a fuel tank and designed to deliver the fuel under high pressure to an injector in accordance with a command from an electronic controller.

One known type of such a fuel pump is generally called a regenerative pump or a Westco type pump. Regarding the Westco type fuel pump, it is known that pump performance such as efficiency is greatly affected by a cross section of a flow passage section and a configuration of vanes of an impeller.

Japanese Patent Publication No. 63-63756, Japanese Utility Model Publication No. 3-2720, or Japanese Patent Laid-Open No. 60-47894, for example, discloses a Westco type fuel pump in which a desired level of performance is achieved by setting dimensions, such as a flow passage representative size Rm, to particular values.

A conventional Westco type fuel pump will be described below with reference to Figs. 21 and 22.

An impeller 9 of the conventional Westco type fuel pump has a disk-like outer configuration. A plurality of vanes 93 and a plurality of vane grooves 92 are provided alternately with equal intervals along both a corner between one side of the disk and its outer peripheral surface and an opposite corner between the other side and the outer peripheral surface. These vanes and vane grooves are positioned on both sides of the impeller 9 with a partition wall 91 therebetween. An outer peripheral surface 910 of the partition wall 91 has a diameter equal to that of an outer peripheral surface 930 of the vane 93. A pump flow passage 95 is defined between an outer periphery of the impeller 9 and an inner surface of a pump casing 90. When the impeller 9 rotates, the outer periphery of the impeller 9 passes through the pump flow passage 95 at a high speed. Therefore, liquid fuel in the vane grooves 92 is given with centrifugal forces to form two vortexes 941, 942 in the pump flow passage 95. With the rotation of the impeller 9, the liquid fuel in the pump flow passage 95 is delivered in a circumferential direction while forming the two vortexes 941, 942. The liquid fuel is subjected to a dynamic pressure to be delivered under high pressure while flowing through the pump flow passage 95.

With the conventional Westco type fuel pump, however, there produces a dead zone 96 between the two vortexes 941, 942 as shown in Fig. 22. In the dead zone 96, the liquid fuel is not given with a sufficient flow speed, thereby causing a counter flow. This raises the problem that the counter flow prevents the fuel from being delivered under high pressure.

To eliminate the counter flow, it could be contemplated to provide a projection radially extending from the casing side or the impeller side in such a manner as to fill up the above dead zone 96. However, providing such a projection to fill up the dead zone 96 gives rise to a fear that the fuel might be partially distributed between both sides of the impeller because of prevention of its movement therebetween.

Westco type pumps are also practiced for uses other than fuel pumps. Japanese Patent Laid-Open No. 61-210288, for example, discloses one of Westco type water pumps. The disclosed technique is intended to suppress the aforesaid counter flow produced in the pump flow passage due to the presence of the dead zone. This prior art proposes that the distal end of the impeller's partition wall is pointed. The disclosed prior art also proposes that the height of the impeller's partition wall is made lower than that of its vanes to position the distal end of the partition wall inside the vanes.

Such a configuration that the height of the impeller's partition wall is made lower than that of its vanes to position the distal end of the partition wall inside the vanes, is also disclosed in Japanese Patent Laid-Open No. 56-32095 relating to an air pump.

However, the water pump disclosed in Japanese Patent Laid-Open No. 61-210288 or the air pump disclosed in Japanese Patent Laid-Open No. 56-32095 is greatly different from a fuel pump in required levels of a delivery capacity under pressure, an impeller diameter and other factors. For that reason, if the disclosed technique relating to the water or air pump is directly applied to a fuel pump, it would be difficult to achieve desired pump performance and operating effect.

A typical water pump, for example, requires a flow rate of 100 to 10000 l/h and a lift of 5 to 10 kgf/cm^2. On the contrary, a typical fuel pump for automobiles requires a flow rate of 50 to 200 l/h and a lift of 2 to 5 kgf/cm^2. Thus, parameter ranges required for practical operation of both the pumps are different from each other to a large extent. Further, an impeller of a water pump is typically about 100 mm in diameter, while an impeller conventionally used in a fuel pump for automobiles is about 50 mm or 30 mm in diameter because the impeller size has limitations from the necessity of being located in an automobile fuel tank.

In addition, an air pump is greatly different from a fuel pump not only in rated values of capacity, efficiency, impeller diameter, etc., but also in such characteristics as compressibility and viscosity of a target substance since a fluid to be pressurized by the air pump is gas. Thus, the air pump disclosed in Japanese Patent Laid-Open No. 56-32095 is formed to have a short radial distance between the vane distal ends of the impeller and the wall surface of the flow passage.

Furthermore, because the impeller diameter is large in water and air pumps, impellers are generally manufactured using metal materials. The metal impeller can be machined to cut the vane grooves for making the distal end of the partition wall pointed. On the contrary, because of a small diameter, the impeller of a fuel pump is generally molded by, for example, injection molding, using resin materials. This means that it is difficult to make the distal end of the partition wall pointed in the fuel pump for the reason of a deformation or cracks often caused when a molding is released from molds. Particularly, a fuel pump having a smaller impeller diameter has the problem that a slight deformation of the impeller configuration affects a fuel flow passing through the flow passage and lowers pump efficiency. Consequently, there is a difficulty in achieving desired pump performance by directly applying the configuration of the conventional water or air pump to a fuel pump.

SUMMARY OF THE INVENTION

In view of the above-mentioned problems encountered in the prior art, an object of the present invention is especially to improve performance of a fuel pump.

To this end, the invention is intended for an improvement of a fuel pump comprising a disk-like impeller which is made of a resin and has vane grooves and vane plates formed alternately along an outer periphery of the impeller, the vane grooves being respectively open to both sides of the impeller and its outer peripheral surface and being parted by a partition wall in an axial direction of the impeller, to define the vane plates, a casing which rotatably accommodates the impeller, defines a pump flow passage along the outer periphery of the impeller and has an intake port and a delivery port both communicating with the pump flow passage, and a motor for driving the impeller to rotate the same. The fuel pump is characterized in that each vane groove of the impeller includes a first groove portion for communicating between one side of the impeller and its outer peripheral surface, a second groove portion for communicating between the other side and the outer peripheral surface of the impeller, and a communicating groove positioned radially outside the first and second groove portions for allowing the first and second groove portions to communicate with each other in the axial direction, the first and second groove portions and the communicating groove being defined between side walls of adjacent twos of the vane plates, and each partition wall is positioned between the first and second groove portions to provide bottom surfaces of the first and second groove portions, the bottom surfaces being formed to gradually approach each other while extending in a radial direction from an inner side toward an outer side of the impeller, and being terminated at a position inside an outer peripheral end of each vane plate with a distance not smaller than a predetermined value between the bottom surfaces to define the communicating groove.

With the fuel pump arangement of the invention summarized above, the impeller has the vane plates and the partition walls which define the respective vane grooves on both sides of the impeller. The partition walls according to the invention are each terminated at a position inside the outer peripheral end of each vane plate such that the opposite bottom surfaces of each vane groove has a distance not smaller than the predetermined value at their outermost ends.

Accordingly, the distal ends of the partition walls are not positioned to directly face the outermost periphery of the impeller and, therefore, vortex flows of fuel generated along the bottom surfaces of each vane groove extend over the entire flow passage and thus reduce the flow dead zone to increase pump efficiency.

Experiments made by the present inventors have proved that by terminating the bottom surface of each vane groove with a distance not smaller than the predetermined value at their outermost ends, there can be obtained higher pump performance than such a configuration of the partition wall that the bottom surfaces are terminated in contact with each other to provide a pointed end.

It is considered that the higher pump performance is resulted because an area into which vortex fuel will not directly enter is formed outside the distal end of each partition wall and the fuel in that area allows the vortex fuel flows to be smoothly merged together outside the distal end of the partition wall.

Further, by terminating the bottom surfaces of each vane groove with a distance not smaller than the predetermined value at their outermost ends, deformation of the distal end of the partition wall at its outermost periphery, which would otherwise occur upon release of a molding from molds, can be prevented, making it possible to obtain then impeller of a desired shape and achieve desired pump

performance with certainty. Thus, the impeller can be manufactured by molds, with the result of improved production efficiency.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a fragmentary perspective view showing an impeller of a fuel pump according to a first embodiment of the present invention;
Fig. 2 is an enlarged sectional view, taken along the II-II line in Fig. 4, showing principal parts of the fuel pump of the first embodiment;
Fig. 3 is a sectional view of the impeller of the first embodiment;
Fig. 4 is a front view of the impeller of the first embodiment;
Fig. 5 is a sectional view of the fuel pump of the first embodiment;
Fig. 6 is a schematic view showing a fuel injection system using the fuel pump of the first embodiment;
Fig. 7 is a fragmentary sectional view of a mold used for molding the impeller of the first embodiment;
Fig. 8 is a graph showing pump efficiency of the fuel pump of the first embodiment when dimension parameters d1 = d2 and L1 are changed;
Fig. 9 is a graph showing pump efficiency of the fuel pump of the first embodiment when a dimension parameter t is changed;
Fig. 10 is a graph showing pump efficiency of a conventional fuel pump when a dimension parameter D is changed;
Fig. 11 is a graph showing pump efficiency of the conventional fuel pump when a dimension parameter k is changed;
Fig. 12 is an enlarged sectional view showing principal parts of an impeller made by way of trial under application of the invention;
Fig. 13 is an enlarged sectional view showing principal parts of an impeller of a first comparative example made by way of trial for comparing performance with the impeller of Fig. 12;
Fig. 14 is an enlarged sectional view showing principal parts of an impeller of a second comparative example;
Fig. 15 is a graph showing pump efficiencies of fuel pumps using the impellers of Figs. 12 to 14;
Fig. 16 is an enlarged sectional view showing principal parts of an impeller of a fuel pump according to a second embodiment of the invention;
Fig. 17 is a fragmentary perspective view showing an impeller of a fuel pump according to a third embodiment of the invention;
Fig. 18 is a fragmentary perspective view showing an impeller of a fuel pump according to a fourth embodiment of the invention;
Fig. 19 is a fragmentary perspective view showing an impeller of a fuel pump according to a fifth embodiment of the invention;
Fig. 20 is a fragmentary plan view showing an impeller of a fuel pump according to a sixth embodiment of the invention;
Fig. 21 is a fragmentary perspective view a conventional impeller; and
Fig. 22 is an enlarged sectional view showing principal parts of the impeller of Fig. 21.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

A fuel pump according to the first embodiment of the invention will be described below with reference to Figs. 1 to 6. The fuel pump is used with a fuel supply system of an internal combustion engine for a motor vehicle.

The entire structure of the fuel pump will be first explained by referring to Figs. 4 and 5.

The fuel pump is comprised of a motor section 2 and a pump section 3.

The motor section 2 comprises a permanent magnet 21 disposed on an inner wall surface of a substantially cylindrical housing 1, and an armature 22 rotatably disposed inside the permanent magnet 21 in concentric relation to the magnet.

The pump section 3 comprises casings 311, 312 fixed to one end of the housing 1, and a disk-like impeller 32 rotating in a disk-shaped space defined between the casings 311 and 312 in concentric relation to the space. The impeller 32 is attached to a shaft 220 of the armature 22 penetrating through the casing 311.

Between an outer periphery of the impeller 32 and the casings 311, 312, there is formed a pump flow passage (hereinafter referred to simply as a flow passage) 33. The flow passage 33 has an intake port 41 at

one end thereof and a delivery port 43 at the other end, and is formed into a C-shape along the outer periphery of the impeller 32. Fuel is introduced to the flow passage 33 through the fuel intake port 41 which is formed in the casing 312.

The flow passage 33 is formed into the C-shape along the outer periphery of the impeller 32, as mentioned above, and has an intake portion 331 and a delivery portion 332 formed in respective predetermined positions with a parting wall 333 therebetween (see Fig. 4). These flow passage intake and delivery portions 331, 332 are larger in radial size than other portions of the flow passage 33, and the flow passage intake portion 331 is larger in radial size than the flow passage delivery portion 332. The flow passage intake portion 331 communicates with the fuel intake port 41, while the flow passage delivery portion 332 communicates with the interior of the housing 1 via the fuel delivery port 43 which is bored to penetrate through the casing 311.

The fuel in the housing 1 is delivered from a fuel delivery portion 42 provided at an opposite end of the housing 1. A connector is provided beside the fuel delivery portion 42 and has a terminal 23 through which electric power is supplied to the motor section 2. The terminal 23 is connected to a brush (not shown) via noise preventing elements such as a coil and a capacitor.

The configuration of the impeller 32 will now be described in detail with reference to Figs. 1 to 4.

The impeller 32 is rotatably accommodated between the casings 311 and 312 which are fixed in the housing 1 by press-fitting.

A plurality of vane plates 323 are formed around the outer periphery of the impeller 32 with predetermined intervals, and a vane groove 322 is formed between adjacent twos of the vane plates 323.

Each vane groove 322 includes groove portions 322a, 322b respectively positioned on both lateral sides of the impeller 32 at its outer periphery, and another groove portion (hereinafter referred to sometimes as a communicating groove) 322c positioned at the outermost periphery of the impeller 32 for communicating the groove portions 322a, 322b with each other in an axial direction. These vane groove portions 322a, 322b, 322c collectively define the vane groove 322 which is substantially C-shaped in cross-section and extends from one lateral side to the other lateral side of the impeller 32 while passing the outermost periphery thereof.

The vane plate 323 is formed between every two vane grooves 322 and 322 adjacent each other in a circumferential direction. Each vane plate 323 has a radial vane shape which extends outwardly perpendicular to the circumferential direction, and is adjacent the vane grooves 322 on its both sides in the circumferential direction to form side walls of the vane grooves 322.

Each pair of vane groove portions 322a, 322b positioned on both sides of the impeller 32 are parted from each other by a partition wall 321 which tapers toward the outermost periphery of the impeller 32. The partition wall 321 has a small flat portion at its distal end and two opposite slopes which define a bottom surface 3221 of the groove portion 322a and a bottom surface 3222 of the groove portion 322b. These bottom surfaces 3221, 3222 are each formed as a curved surface having the radius of curvature R (see Fig. 2). The axial distance between the bottom surface 3221 and the bottom surface 3222 is gradually reduced toward the outermost periphery of the impeller 32 to become minimum at the outermost end of the partition wall 321. This minimum distance is also determined as a distance between the outermost ends of the bottom surfaces 3221 and 3222. Further, an outer peripheral surface 3210 of the partition wall 321 defines a bottom surface of the vane groove portion 322c.

With the above structure, the fuel is urged in a rotating direction of the impeller 32 by not only side walls of the vane groove portions 322a, 322b, but also side walls of the vane groove portions (communicating grooves) 322c.

The partition wall 321 of the impeller 32 is arranged, as shown in Figs. 1 and 2, such that the outer peripheral surface 3210 is located radially inside outer peripheral surfaces 3230 of the vane plates 323 which define the outermost peripheral surface of the impeller 32. In this embodiment, the radial entire length L1 of each vane groove portion 322c, i.e., the radial distance between the outer peripheral surface 3210 of the partition wall 321 and the outer peripheral surface 3230 of the vane plate 323, is set to 40 % of the length L2 of each vane plate 323 (see Fig. 2).

As shown in Figs. 3 and 4, the vane plates 323 and the vane grooves 322 are disposed alternately with predetermined intervals around the outer periphery of the impeller 32 in the circumferential direction. At the center of the impeller 32, there is thoroughly bored a shaft hole 325 for allowing the shaft 220 to be fitted into and penetrate through the hole 325.

Various dimensions or sizes of the impeller in the embodiment described above are as shown in Table 1 below.

Table 1

D	t	d1 d2	d3	R	L1	L2	Rm	k
30	2.4	0.7	0.7	4	1.0	2.4	0.7	0.3
D: diameter t: thickness d1, d2: axial gap d3: radial gap R: curvature of recessed surface L1: entire radial length of communicating passage L2: entire radial length of vane Rm: flow passage representative size k: end face length of partition wall								

In Table 1, the diameter D indicates a diameter of the impeller including the vanes at the outer periphery; the thickness t indicates an axial thickness of the impeller; the axial gap d1 = d2 indicates a distance between axial ends of each vane plate 323 and inner lateral surfaces of the casings 311, 312; and the radial gap d3 indicates a distance between a radial end of each vane plate 323 and an inner peripheral surface of the casing 311. The curvature of recessed surfaces R indicates a radius of curvature of both the sloped bottom surfaces of each partition wall 321 of the impeller; the entire radial length of communicating passage L1 indicates a radial length of the communicating passage or groove 322C from the outer peripheral surface 3210 of each partition wall 321 to the outer peripheral surface 3230 of each vane plate 323; and the entire radial length of vane L2 indicates a radial length of each vane plate 323 from its inner periphery to its outer peripheral surface 3230, including the communicating passage. The flow passage representative size Rm is determined by S/l on the assumption that the axial sectional area of the flow passage defined by segments of a - b - c - d - j - i - h - g - f - e - a in Fig. 2 is S and the peripheral length of a section along peripheral edges of the impeller defined by segments of a - b - c - d in Fig. 2. The end face length of partition wall k indicates an axial length of the outer peripheral surface 3210 of each partition wall 321. The values shown in Table 1 are in the unit of mm.

As shown in Fig. 6, the fuel pump of this embodiment is installed in a fuel tank 61 which is mounted on a motor vehicle, and is connected to an onboard battery 62. Then, the fuel pump supplies fuel 63 in the fuel tank 61 to a fuel injection system 64. A fuel filter 65 is connected to the fuel intake port 41 of the fuel pump, and a piping line 66 is connected to the fuel delivery port 42. The piping line 66 supplies the fuel to injectors 67 of the fuel injection system 64, and the fuel pressure is adjusted by a regulator 68 to a predetermined value. The fuel discharged from the regulator 68 is returned to the fuel tank 61 again via a return piping line 69. Each of the injectors 67 sprays the fuel into an intake passage of an engine 70.

The fuel pump which is used with the fuel injection system of an internal combustion engine for a motor vehicle like this embodiment is operated on the condition that a delivery rate is in the range of 50 to 200 l/h and a lift is in the range of 2 to 5 kgf/cm^3. Taking into account environmental conditions under which the motor vehicle is used, the fuel pump is designed to operate in the temperature range of about - 30 to 80 °C without any troubles.

For the fuel pump used under the condition of the above delivery rate and delivery pressure, preferably, the impeller diameter is set to 20 - 65 mm and the flow passage representative size Rm is set to 0.4 - 2.0 mm. More preferably, the flow passage representative size Rm is set to 0.6 - 1.6 mm. Specific dimensions of such a fuel pump are disclosed in Japanese Patent Publication No. 63-63756 or U.S. Patent No. 4,493,620.

A description will now be made on a process of manufacturing the impeller of this embodiment. Fig. 7 is a fragmentary sectional view of a mold used for molding the impeller 32. Fig. 7 shows a section of the part corresponding to the vane groove 322.

A mold 72 comprises two portions 74, 75 divided at a mold parting plane 73 which corresponds to the axial center of the impeller 32. The inner cavity configuration of the mold 72 is formed in accordance with the shape of the impeller 32, though that the cavity is slightly larger than the impeller 32 in directions of its diameter and thickness. In Fig. 7, a broken line 76 indicates the inner cavity configuration of the mold 72, and a two-dot-chain line 77 indicates the final shape of the impeller 32. As will be seen from Fig. 7, the inner cavity configuration of the mold 72 at a position corresponding to the vane groove 322 of the impeller 32 is similar to the shape of the impeller 32.

To manufacture the impeller 32, a thermosetting resin is first poured into the mold 72 for roughly molding the outer configuration of the impeller. The molded impeller is somewhat larger in diameter and thickness than the finished impeller 32. The molded impeller has the same configuration in its portion corresponding to each vane groove 322 as the finished impeller 32. The material of the impeller 32 is a phenol resin mixed with glass fibers as reinforcements. Use of such a thermosetting resin lessens volume changes due to temperature changes and enables the pump to operate while maintaining a high level of

performance over a wide range from low to high temperatures.

Then, both lateral sides and an outer peripheral sides of the impeller molded by using the mold 72 are grounded. Specifically, in this grinding step, both the lateral sides of the impeller, as well as the outer peripheral surface 3230 and both side surfaces 3231, 3232 of each vane plate 323 are grounded. After the grinding, the configuration of the impeller 32 as shown in Figs. 1 to 4 is completed. Thus, among the surfaces of the impeller 32 shown in Fig. 1, those surfaces which are indicated by dot patterns are formed by only molding without being ground. Particularly, in this embodiment, the outer peripheral surface 3210 at the distal end of each partition wall 321 is not ground.

As described above, in this embodiment, the impeller 32 is molded by using the mold 72. Therefore, the many vane grooves 322 can be simply formed, making the impeller well adapted for mass production. If the distal end of the partition wall 321 is too thin, it may be deformed when the molds 74 and 75 are opened to release the molding, thus leading to a large influence upon the pump performance. On the contrary, in this embodiment, the outer peripheral surface 3210 of each partition wall 321 is formed into a flat shape to ensure a sufficient thickness even at the distal end of the partition wall 321. Therefore, when the molded impeller is released from and take out of the molds 74 and 75, deformation of the partition wall 321 is prevented. Particularly, in the case of using a thermosetting resin as with this embodiment, there is a fear that the partition wall 321 may crack upon opening the molds because the thermosetting resin is generally brittle. However, by making the distal end of the partition wall 321 thick to increase the strength like this embodiment, the thermosetting resin can be prevented from cracking at the distal end of the partition wall 321.

Operation and advantages of the fuel pump of this embodiment constructed as above will be described below.

When electric power is supplied to the motor section 2 from the battery 63 via the terminal 23, the armature 22 is rotated in the motor section 2. The rotation of the armature 22 is transmitted to the impeller 32 via the shaft 220 for rotating the impeller 32.

With the rotation of the impeller 32, the fuel in the fuel tank 61 is sucked into the flow passage 33 through the fuel intake port 41 and pressurized in the flow passage 33 by the vane plates 323 of the impeller 33. Then, the fuel reaches the fuel delivery port 42 and is delivered under pressure from the fuel delivery port 42 to the injectors 67.

Describing in more detail, when the impeller 32 is rotated in the casings 311, 312, the outer peripheral portion of the impeller 32 passes the flow passage 33 at a high speed. The liquid fuel in the flow passage 33 is moved in the circumferential direction while forming two vortexes 341, 342 due to centrifugal forces, thereby mainly increasing a dynamic pressure of the fuel.

In this embodiment, the flow passage 33 is axially divided by only the partition walls 321 of the impeller 32. Since portions of each vane groove 322 which locate outside the outer peripheral surface 3210 of the partition wall 321 thoroughly communicates with each other in the axial direction via the vane groove portion 322c, the fuel can easily move between the opposite lateral sides of the impeller 32 from one to the other so that the fuel is prevented from being locally distributed in either one lateral side of the impeller. As a result, generation of a pressure axially urging the impeller 32 is suppressed to reduce a friction resistance and noises during the rotation of the impeller 32.

Further, since the outer axial surface of each vane plate 323 which is effective to urge the fuel in the circumferential direction is provided outside the outer peripheral surface 3210 of the partition wall 321, the fuel can be urged circumferentially by the effective outer axial surfaces of the vane plates 323, allowing a larger quantity of fuel to be moved in the rotating direction of the impeller 32. Additionally, the counter vortex which has been conventionally produced between the outer peripheral surface of the partition wall and the inner peripheral surface of the casing can be diminished to enhance a capability of raising the fuel pressure and provide a higher delivery pressure under the same electric power supplied.

Furthermore, in this embodiment, the configuration of only the impeller 32 is changed, while the configurations of the casings and other components remain the same as those of fuel pumps which have been conventionally put into practice. Accordingly, pump performance of the conventional fuel pumps can be greatly improved just by updating facilities for manufacture of impellers, with the result of a quite big practical value.

In order to determine the dimensions of the fuel pump of this embodiment, several experimental pumps were fabricated and their efficiencies were tested. The results of the experiments will be described below for showing that the above-mentioned dimension values of this embodiment can provide good pump efficiency.

In the experiments, for determining the efficiency, the pump input was calculated as the product of a load torque and a rotational speed, and the pump output was calculated as the product of a delivery

pressure and a delivery rate. The delivery pressure was measured by using both a digital multimeter manufactured by Advantest Co. and a small-sized semiconductor pressure sensor manufactured by Toyoda Machine Works Ltd. The delivery flow rate was measured by using a digital flowmeter manufactured by Ono Measuring Instrument Ltd.

First, fuel pumps in combination of plural impellers and different flow passage configurations as shown in Table 2 below were fabricated by way of trial and their efficiencies were measured. The pump efficiencies resulted from these experimental examples are plotted in Fig. 8.

Table 2

No.	D	t	d1 d2	d3	R	L1	L2	Rm	k
1	30	2.4	0.6	0.7	4	0	2.4	0.63	0.3
2	30	2.4	0.7	0.7	4	0	2.4	0.70	0.3
3	30	2.4	0.8	0.7	4	0	2.4	0.77	0.3
4	30	2.4	0.9	0.7	4	0	2.4	0.83	0.3
5	30	2.4	0.7	0.7	4	0.5	2.4	0.7	0.3
6	30	2.4	0.6	0.7	4	1.0	2.4	0.63	0.3
7	30	2.4	0.7	0.7	4	1.0	2.4	0.70	0.3
8	30	2.4	0.8	0.7	4	1.0	2.4	0.77	0.3
9	30	2.4	0.9	0.7	4	1.0	2.4	0.83	0.3
10	30	2.4	0.6	0.7	4	1.5	2.4	0.63	0.3
11	30	2.4	0.7	0.7	4	1.5	2.4	0.70	0.3
12	30	2.4	0.8	0.7	4	1.5	2.4	0.77	0.3
13	30	2.4	0.9	0.7	4	1.5	2.4	0.83	0.3

D: diameter t: thickness d1, d2: axial gap d3: radial gap R: curvature of recessed surface L1: entire radial length of communicating passage L2: entire radial length of vane Rm: flow passage representative size k: end face length of partition wall

In Table 2, the experimental examples Nos. 1 to 4 employ conventional impellers formed with no communicating passages or grooves.

From the graph of Fig. 8 indicating the pump efficiencies of the experimental examples in Table 2, it has proved that the experimental example No. 7 provides the maximum efficiency which is about 35 %.

Of the experimental examples representing the pumps which employ the conventional impellers, No. 2 provides the highest efficiency of about 30 %. It is thus found that by setting the ratio of L1/L2 to about 0.4, any pumps having the communicating passages can provide higher efficiency than the conventional pumps no matter which value d1 = d2 takes in the range of 0.6 to 0.9. It is also found that by setting d1 = d2 in the range of about 0.7 to 0.8, any present pumps can provide higher efficiency than the conventional pumps no matter which value the ratio of L1/L2 takes in the wide range of 0 to about 0.6.

In the characteristic curves plotted in Fig. 8, the efficiencies decline in almost similar manner when the ratio of L1/L2 becomes larger. This is believably due to that since the dimension L1 is changed in the experimental examples, the sloped surfaces of each partition wall become too short to satisfactorily produce the vortexes 341, 342 when the ratio of L1/L2 takes a large value.

Also, in the characteristic curves of Fig. 8, the efficiencies decline in almost similar fashion when the ratio of L1/L2 becomes smaller. This is believably due to that since the radial distal end face of the partition wall 321 approaches the outer periphery of the flow passage, the flow dead zone 96 (see Fig. 22) is generated and the counter flow passing the dead zone 96 reduces the pump efficiencies.

In addition, when the axial gap d1 = d2 is set to 0.6 or 0.9, the pump efficiencies drop to a large extent. This is believably due to that the small axial gap d1 = d2 reduces the delivery rate and also disables generation of the satisfactory vortexes 341, 342.

On the other hand, it is believed that the large axial gap d1 = d2 makes the flow passage too large and hence generates undesired vortexes which lower the pump efficiency.

Further, fuel pumps in combination of plural impellers and different flow passage configurations as shown in Table 3 below were fabricated by way of trial and their efficiencies were measured. The pump efficiencies resulted from these experimental examples are plotted in Fig. 9.

Table 3

No.	D	t	d1 d2	d3	R	L1	L2	Rm	k
21	30	2.0	0.7	0.7	4	0	2.4	0.7	0.3
22	30	2.4	0.7	0.7	4	0	2.4	0.7	0.3
23	30	3.0	0.7	0.7	4	0	2.4	0.7	0.3
24	30	2.0	0.7	0.7	4	0.5	2.4	0.7	0.3
25	30	2.4	0.7	0.7	4	0.5	2.4	0.7	0.3
26	30	3.0	0.7	0.7	4	0.5	2.4	0.7	0.3
27	30	2.0	0.7	0.7	4	1.0	2.4	0.7	0.3
28	30	2.4	0.7	0.7	4	1.0	2.4	0.7	0.3
29	30	3.0	0.7	0.7	4	1.0	2.4	0.7	0.3
30	30	2.0	0.7	0.7	4	1.5	2.4	0.7	0.3
31	30	2.4	0.7	0.7	4	1.5	2.4	0.7	0.3
32	30	3.0	0.7	0.7	4	1.5	2.4	0.7	0.3

D: diameter t: thickness d1, d2: axial gap d3: radial gap R: curvature of recessed surface L1: entire radial length of communicating passage L2: entire radial length of vane Rm: flow passage representative size k: end face length of partition wall

In the experimental examples of Table 3, the entire axial length of the flow passage is changed depending on changes in the thickness t of the impeller.

In Table 3, the experimental examples No. 21 to 23 employ conventional impellers formed with no communicating passages or grooves.

From the graph of Fig. 9 indicating the pump efficiencies of the experimental examples in Table 3, it has proved that the experimental example No. 28 provides the maximum efficiency which is about 35 %.

Of the experimental examples representing the pumps which employ the conventional impellers, No. 22 provides the highest efficiency of about 30 %. It is thus found that by setting the ratio of L1/L2 to about 0.4, any pumps having the communicating passages can provide efficiency almost equal to or higher than the conventional pumps no matter which value the thickness t takes in the range of 2.0 to 3.0.

It is also found that by setting the thickness t in the range of about 2.4 to 3.0, any present pumps can provide efficiency almost equal to or higher than the conventional pumps no matter which value the ratio of L1/L2 takes in the wide range of 0.1 to about 0.6.

In the characteristic curves plotted in Fig. 9, the efficiencies decline in almost similar fashion when the ratio of L1/L2 becomes larger. This is believably due to that since the dimension L1 is changed in the experimental examples, the sloped surfaces of each partition wall, or the bottom of each vane groove, become too short to satisfactorily produce the vortexes 341, 342 when the ratio of L1/L2 takes a large value.

In addition, when the impeller thickness t is set to a small value, the pump efficiencies drop to a large extent. This is believably due to that the small thickness t reduces the vane area of the impeller to such an extent as to disable generation of the satisfactory vortexes 341, 342. Another reason is that since the entire axial length of the flow passage is changed depending on changes in the thickness of the impeller in the above experimental examples, the flow passage becomes too short in the axial direction as a whole to generate the satisfactory vortexes 341, 342.

On the other hand, when the impeller thickness t is set to a large value, the pump efficiencies drop to a small extent. This is believably due to that since the entire axial length of the flow passage is changed depending on changes in the thickness of the impeller in the above experimental examples, the flow

passage becomes too long in the axial direction as a whole to generate the satisfactory vortexes 341, 342.

As described above, upon reviewing the test results of Figs. 8 and 9 obtained from the experimental examples in Tables 2 and 3, it has proved that by providing the communicating passages in the impeller and setting the ratio of L1/L2 in the range of about 0.1 to 0.6, there can be obtained pump efficiency almost equal to or higher than the conventional pumps.

It is also found from the test results of Figs. 8 and 9 that the pump efficiency is maximized at the dimension values of Table 1 when the ratio of L/L2 is set to 0.4. In view of the above test results, the dimension values shown in Table 1 are adopted in this first embodiment.

With the first embodiment, as described above, since the partition walls 321 are more recessed radially inwardly as compared with the conventional impellers, the vortexes generated along the opposite sloped surfaces of each partition wall 321 can flow into the dead zone 96 (see Fig. 22) of the flow passage which has been formed in the conventional pumps, so that generation of the counter flow in the dead zone 96 is prevented to improve the pump efficiency.

Description will no be given of the reason why the diameter D of the impeller and the axial length k of the outer end face of the partition wall are set in the first embodiment to the values shown above in Table 1.

Fig. 10 is a graph showing pump efficiencies resulted when the diameter D of the impeller is changed in the conventional Westco type fuel pump having no communicating groove.

A characteristic curve plotted in Fig. 10 was obtained by fabricating plural pumps by way of trial which had dimension values as shown in Table 4 below with only the diameter D of the impeller changed, and measuring their pump efficiencies.

Table 4

D	t	d1 d2	d3	R	L1	L2	Rm	k
-	2.4	0.7	0.7	4	0	2.4	0.7	0.3
D: diameter t: thickness d1, d2: axial gap d3: radial gap R: curvature of recessed surface L1: entire radial length of communicating passage L2: entire radial length of vane Rm: flow passage representative size k: end face length of partition wall								

From Fig. 10, it is found that, in spite of the conventional pumps, the efficiency at an almost satisfactory level not less than 20 % can be obtained when the impeller diameter is in the range of 20 mm to 65 mm. It is also presumed that in the case of the present pumps as well which have the communicating grooves like the above first embodiment and hence the different impeller configuration, the efficiency at an almost satisfactory level can be obtained with the dimension values close to those shown in Table 4 except the impeller diameter D, when the impeller diameter is in the range of 20 mm to 65 mm.

Additionally, in the characteristic curve of Fig. 10, if the impeller diameter D is set below 20 mm, the efficiency declines to a large extent and, if the impeller diameter D is set above 65 mm, the efficiency declines gently. This is believably due to that the small impeller diameter makes the length of the flow passage too short to provide such passage portions as effectively functioning as a pump. It is also believed that the large impeller diameter makes a sliding resistance due to warping of the impeller so large as to lower the efficiency.

Fig. 11 is a graph showing pump efficiencies resulted when the axial length k of the outer peripheral surface (radial distal end face) of the impeller partition wall is changed in the conventional Westco type fuel pump having no communicating groove.

A characteristic curve plotted in Fig. 11 was obtained by fabricating plural pumps by way of trial which had dimension values as shown in Table 5 below with only the aforesaid length k changed, and measuring their pump efficiencies.

Table 5

D	t	d1 d2	d3	R	L1	L2	Rm	k
30	2.4	0.7	0.7	4	0	2.4	0.7	-
D: diameter t: thickness d1, d2: axial gap d3: radial gap R: curvature of recessed surface L1: entire radial length of communicating passage L2: entire radial length of vane Rm: flow passage representative size k: end face length of partition wall								

From Fig. 11, it is found that, in spite of the conventional pumps, the efficiency at an almost satisfactory level not less than 20 % can be obtained when the length k is in the range of 0.3 mm to 0.8 mm. Judging from the characteristic curve of Fig. 11, it is further presumed that the efficiency at an almost satisfactory level not less than 20 % would be also obtained when the length k is in the range not larger than 0.3 mm. However, since the impeller is formed of resin materials and molded by using molds as shown in Fig. 7, setting the length k below 0.2 mm is difficult from the standpoints of strength and feasibility of the manufacture technique.

From the above test results, it is also presumed that in the case of the present pumps as well which have the communicating grooves like the above first embodiment and hence the different impeller configuration, the efficiency at an almost satisfactory level can be obtained with the dimension values close to those shown in Table 5 except the length k, when the length k is in the range of 0.2 mm to 0.8 mm.

Additionally, in the characteristic curve of Fig. 11, if the length k is set above 0.8 mm, the efficiency declines gently. This is believably due to that the large length k increases the flow dead zone 96 (see Fig. 22) excessively.

From the results of the above experiments, it is believed that with the present pumps having the communicating grooves like the first embodiment, high efficiency can be obtained when the ratio of L1/L2 is in the range of about 0.1 to 0.6. It is also believed that high efficiency can be obtained by setting the ratio of L1/L2 in the range of about 0.1 to 0.6 when the axial gap d1 = d2 is in the range of 0.7 mm to 0.8 mm. Further, it is believed that high efficiency can be obtained by setting the ratio of L1/L2 in the range of about 0.1 to 0.6 when the impeller thickness t is in the range of 2.4 mm to 3.0 mm. In addition, it is presumed that the above operating advantage is obtained when the impeller diameter D is in the range of about 20 mm to 65 mm, and when the axial length k of the outer peripheral surface of the partition wall is in the range of about 0.2 mm to 0.8 mm.

Description will next be given of a specific advantage due to the outer peripheral surface 3210 of the partition wall. In the following, test results will be explained, which were obtained by fabricating impellers having respective configurations shown in Figs. 12, 13 and 14. All these experimental impellers have vane grooves which were formed by cutting outer peripheral portions of disk plates.

First, Fig. 12 is a sectional view of the impeller 32 and the flow passage of the test example to which the present invention is applied. The relevant dimensional values are as shown in Table 6 below.

Table 6

D	t	d1,d2 d3	R	L3	L1	L2	Rm	k
30	2.35	0.7	4	0.6	1.0	2.4	0.7	0.3
D: diameter t: thickness d1, d2; d3: axial and radial gap R: curvature of recessed surface L3: distance to imaginary cross point L1: entire radial length of communicating passage L2: entire radial length of vane Rm: flow passage representative size k: end face length of partition wall								

The impeller 32 of Fig. 12 is of the same configuration as the impeller 32 described before by referring to Fig. 2 and Table 1. Note that the thickness t in Table 1 is given as 2.4 mm by rounding 2.35 mm to one decimal place.

In Fig. 12, V1 represents an imaginary cross point at which the bottom surfaces 3221, 3222 of the vane groove 322 would intersect with each other when extended with their radius of curvature. In this test example, the imaginary cross point V1 locates in the communicating passage or groove 322c nearly at the middle of the entire radial length L1 of the communicating passage.

Fig. 13 is a sectional view showing the configurations of an impeller and a flow passage of a first comparative example.

In this first comparative example, under the same dimension values as shown in Table 6, bottom surfaces 131, 132 of each vane groove are formed so as to intersect with each other inside the distal end of each vane plate by moving their centers of curvature. Accordingly, the dimension values of this example are as shown in Table 7 below. The distance to cross point L3 indicates a distance between the distal end of each vane plate and an distal end 133 of each partition wall.

Table 7

D	t	d1,d2 d3	R	L3	L1	L2	Rm	k
30	2.35	0.7	4	1.0	1.0	2.4	0.7	0.3
D: diameter t: thickness d1, d2; d3: axial and radial gap R: curvature of recessed surface L3: distance to cross point L1: entire radial length of communicating passage L2: entire radial length of vane Rm: flow passage representative size k: end face length of partition wall								

Fig. 14 is a sectional view showing the configurations of an impeller and a flow passage of a second comparative example.

In this second comparative example, under the same dimension values as shown in Table 6, bottom surfaces of each vane groove are formed so as to intersect with each other at the distal end of each vane plate by moving their centers of curvature. Accordingly, the dimension values of this example are as shown in Table 8 below.

Table 8

D	t	d1,d2 d3	R	L3	L1	L2	Rm	k
30	2.35	0.7	4	0	0	2.4	0.7	0
D: diameter t: thickness d1, d2; d3: axial and radial gap R: curvature of recessed surface L3: distance to cross point L1: entire radial length of communicating passage L2: entire radial length of vane Rm: flow passage representative size k: end face length of partition wall								

Fig. 15 is a graph showing pump efficiencies of the test example, the first comparative example, and the second comparative example respectively illustrated in Figs. 12, 13 and 14. In Fig. 15, a solid line plotted by squares represents the test example of Fig. 12, a broken line plotted by triangles represents the first comparative example of Fig. 13, and a one-dot-chain line plotted by circles represents the second comparative example of Fig. 14.

As will be seen from the graph of Fig. 15, the impeller having the configuration of Fig. 12 has proved to be maximum in efficiency. It is believed that the resulting difference in efficiency is attributable to difference in vortex flows of fuel caused by the rotation of the impeller.

In the case of Fig. 12, the vortex fuel flows generated around each vane groove first flow along the bottom surfaces 3221, 3222 of the vane groove and then merge together near the center of the communicating passage to flow outwardly in the radial direction. In this test example, the (outer peripheral) distal end surface 3210 of each partition wall of the impeller is formed into a flat surface with a predetermined thickness or length k. With this configuration, there is formed outside the distal end surface 3210 of the partition wall an area into which the vortex fuel flows coming along the bottom surfaces 3221, 3222 will not directly enter, and the fuel stagnates in that area. It is considered that the higher pump efficiency as shown in Fig. 15 is resulted because the fuel stagnating outside the distal end surface 3210 of the partition wall acts to allow the vortex fuel flows coming along the bottom surfaces 3221, 3222 to smoothly merge together.

Meanwhile, in the case of the first comparative example of Fig. 13, the vortex fuel flows coming along the bottom surfaces 3221, 3222 abruptly strike against each other just outside the pointed distal end 133 of the partition wall, and then flow outwardly in the radial direction. With the configuration as shown in Fig. 13, however, when the fuel currents flows vortical along the bottom surfaces 3221, 3222 strike against each other, both the fuel flows have large axial components of flow speeds. It is believed that such axial speed

components make both the fuel flows dampen mutually and hence so weaker as to provide the pump efficiency lower than in the case of the configuration of Fig. 12.

For moderating the above mutual collision of the fuel flows, it could be contemplated to make smaller an angle α indicated in Fig. 13. However, when the impeller is molded using resin materials, there is a difficulty in realizing the desired small angle α from the standpoints of strength and feasibility of the manufacture technique.

For the case of the second comparative example of Fig. 14, it is believed that the mutual collision of the fuel flows are moderated in comparison with the example of Fig. 13, but the vertically long partition wall makes so small the volume of the vane groove as to reduce the pump efficiency.

In sum, it is believed that with the impeller of Fig. 12, the mutual collision of the two vortex fuel flows are prevented by the fuel stagnating outside the distal end of the partition wall, that the stagnating fuel functions as an imaginary partition wall which allows the two vortex fuel flows to smoothly merge together. As a result, the strong vortex fuel flows are produced over the region from the vane grooves of the impeller to the flow passage, thereby providing a high level of pump efficiency.

Fig. 16 shows a sectional view of an impeller of a fuel pump according to a second embodiment of the invention.

In the impeller of this second embodiment, as with the first embodiment, bottom surfaces of each vane groove are formed with the radius of curvature corresponding to the vortex fuel flows generated by the rotation of the impeller. A predetermined thickness k is ensured between outermost peripheral ends of the bottom surfaces which have the radius of curvature corresponding to the vortex fuel flows. More specifically, in this second embodiment, a curved surface 163 is formed at a distal end of each partition wall 162 of the impeller 161. The partition wall 162 of this second embodiment has the thickness k of about 0.3 mm between outermost peripheral ends 164a, 165a of bottom surfaces 164, 165 having the radius of curvature R. In other words, in this second embodiment, the thickness k of the partition wall 162 is set to about 0.3 mm at flexion points of curved lines which define an outer configuration of the partition wall 162. Dimension values of other pump components are the same as those in the first embodiment. Also in this embodiment, there is formed outside the curved surface at the distal end of the partition wall an area into which the vortex fuel flows coming along the bottom surfaces 164, 165 will not directly enter. Then, the fuel stagnating in that area acts to allow the two vortex fuel flows to smoothly merge together.

Fig. 17 shows a perspective view of an impeller of a fuel pump according to a third embodiment of the invention.

In the impeller of this third embodiment, an upper rear edge of each vane plate 323, i.e., an upper end corner of each vane plate 323 on its trailing side with respect to the rotating direction of an impeller 32, is slantly chamfered to form a sloped surface 3231a. The fuel flowing out of one vane groove 322 of the impeller 32 in the form of the aforesaid vortexes 341, 342 is introduced, after swirling over the flow passage, into another succeeding vane groove 322 again to be given with further vortex forces. At this time, with the impeller of this third embodiment, the vortexes 341, 342 flowing out of the one vane groove 322 are more easily introduced to the succeeding vane groove 322. Accordingly, loss of the vortex fuel flows generated by the impeller 32 is reduced to raise the pump efficiency.

Fig. 18 shows a perspective view of an impeller of a fuel pump according to a fourth embodiment of the invention.

In the impeller of this fourth embodiment, in addition to the sloped surface 3231a in Fig. 17, an upper front edge of each vane plate 323 is also slantly chamfered to form a sloped surface 3231b. Accordingly, loss of the vortex fuel flows is reduced as with the above third embodiment. Moreover, this fourth embodiment has another advantage that the impeller 32 can be assembled without taking into account the rotating direction.

Fig. 19 shows a perspective view of an impeller of a fuel pump according to a fifth embodiment of the invention.

An impeller 32 of this fifth embodiment is different from that of the above fourth embodiment shown in Fig. 18 in that the front sloped surface 3231b is chamfered to a smaller extent and the rear sloped surface 3231a is chamfered to a larger extent, thus making both the sloped surfaces asymmetrical in their shapes. This fifth embodiment can also reduce loss of the vortex fuel flows similarly to the above embodiments.

Fig. 20 shows a plan view of an impeller of a fuel pump according to a sixth embodiment of the invention.

In this sixth embodiment, an impeller 32 is formed such that each partition wall 321 is jointed to two adjacent vane plates 323 through smooth curved portions 3214, 3215 at both circumferential ends of an outer peripheral surface 3210 of the partition wall 321. With this embodiment, a circumferential fuel current following the outer peripheral surface 3210 of the partition wall 321 flows along the curved portions 3214,

3215 in joint areas with the vane plates 323. Therefore, the fuel current is not impeded and its loss can be reduced. Additionally, at the time when grinding the outer periphery of the impeller, a large stress is exerted on the vane plate 323. By providing the curved portions in the joint areas with the vane plates 323 like this sixth embodiment, the vane plates 323 can be so reinforced as to prevent possible damage or deformation of the vane plates 323.

It should be noted that in the figures referred to for description of the foregoing embodiments, the configuration of the impeller in each embodiment is exaggerated and, in particular, the distal end face of the partition wall of the impeller is illustrated to be larger than the actual dimensions.

Further, the impeller of each fuel pump is supposed to rotate at a rotational speed of 3000 to 15000 rpm as usual. Thus, in the foregoing embodiments, desired pump performance is obtained in such a range of the rotational speed.

Although the invention has been described in conjunction with the embodiments, it should be understood that the invention can be practiced in various forms other than the illustrated specific forms without departing from the scope of the attached claims.

A Westco type fuel pump includes an impeller (32) which has a plurality of vane grooves (322) and a plurality of vane plates (323) provided alternately along its outer periphery. Each vane groove (322) is constituted by groove portions (322a, 322b) formed in both sides of the impeller (32), respectively, with a partition wall (321) provided between the groove portions (322a, 322b). The partition wall has an outer peripheral surface (3210) located radially inside an outer peripheral surfaces (3230) of each vane plate (323) and has a predetermined thickness in an axial direction of the impeller. As the impeller (32) rotates, two vortex flows of fuel are generated along bottom surfaces (3221, 3222) of the groove portions (322a, 322b) and then smoothly merge together at a position outside the outer peripheral surface (3210) of the partition wall, thereby reducing a flow dead zone (96) in a pump flow passage (33). When the impeller (32) is molded by using molds, deformation of the molded impeller is prevented due to the thickness of the outer peripheral surface (3210). Of the surfaces of the impeller (32), therefore, the surfaces of each vane groove remain as they are after the molding, while both sides of the impeller (32) and the outer peripheral surface (3230) of the vane plates (323) are ground. Thus, the impeller (32) able to surely achieve a high level of pump performance can be easily provided by resin molding.

Claims

1. A fuel pump comprising a disk-like impeller which is made of a resin and has vane grooves and vane plates formed alternately along an outer periphery of the impeller, the vane grooves being respectively open to both lateral sides of the impeller and its outer peripheral surface and being parted by a partition wall in an axial direction of the impeller to define the plates, a casing which rotatably accommodates the impeller, defines a pump flow passage along the outer periphery of the impeller and has an intake port and a delivery port both communicating with the pump flow passage, and a motor for driving the impeller to rotate the same, characterized in that

   each vane groove (322) of said impeller (32) includes a first groove portion (322a) for communicating between one lateral side and said outer peripheral surface of said impeller, a second groove portion (322b) for communicating between another lateral side and said outer peripheral surface of said impeller, and a communicating groove (322c) positioned radially outside said first and second groove portions for allowing said first and second groove portions to communicate with each other in an axial direction, said first and second groove portions and said communicating groove being defined between side walls of adjacent twos of vane plates (323), and each partition wall (321) is positioned between said first and second groove portions to provide bottom surfaces (3221, 3222) of said first and second groove portions, said bottom surfaces being formed to gradually approach each other while extending in a radial direction from an inner side toward an outer side of said impeller, and being terminated at a position inside an outer peripheral end (3230) of each vane plate with a distance (k) not smaller than a predetermined value between said bottom surfaces to define said communicating groove (322c).

2. A fuel pump according to claim 1, characterized in that each of said partition walls (321) has at its outer periphery a distal end face for joining said bottom surface (3221) of said first groove portion (322a) and said bottom surface (3222) of said second groove portion (322b) to each other.

3. A fuel pump according to claim 2, characterized in that said distal end face is a flat surface (3210).

4. A fuel pump according to claim 2, characterized in that said distal end face is a curved surface (163).

5. A fuel pump according to claim 2, characterized in that said impeller (32) is formed by molding such that the distal end face (3210, 163) of each said partition wall, as well as said bottom surfaces (3221, 3222) of said first and second groove portions and side surfaces of said first and second groove portions and said communicating groove (322c) remain as they are after the molding, whereas said outer peripheral surface (3230) and axial lateral surfaces (3231, 3232) of each said vane plate are ground after the molding.

6. A fuel pump according to any one of claims 1 to 5, characterized in that the fuel pump is installed in a fuel tank (63) to feed fuel to a fuel injection system (64) of an internal combustion engine (70) and is used with a delivery pressure in the range of 2 to 5 kgf/cm^2 and a delivery rate in the range of 5 to 200 l/h, a diameter (D) of said impeller (32) is in the range of 20 to 65 mm, and a flow passage representative size (Rm) defined by said impeller and said flow passage is in the range of 0.4 to 2.0 mm.

7. A fuel pump according to claim 6, characterized in that a ratio L1/L2 of a distance L1 between said outer peripheral end (3230) of each said vane plate (323) and said distal end of each said partition wall (321) to an entire length L2 of each said vane plate is in the range of 0.1 to 0.6.

8. A fuel pump according to claim 7, characterized in that said bottom surfaces (3221, 3222) are terminated at their outermost peripheries with a distance in the range of 0.2 to 0.8 mm left therebetween.

9. A fuel pump according to claim 1, characterized in that each of said vane plates (323) has a surface (3231a) sloped in a rotating direction of said impeller (32).

10. A fuel pump according to claim 2, characterized in that joint portions (3214, 3215) between each said partition wall (321) and adjacent vane plates (323) are formed smoothly.

11. A fuel pump comprising a disk-like impeller which is made of a resin and has vane plates and partition walls provided alternately along an outer periphery of the impeller, said vane plates being each formed to face in a circumferential direction of the impeller, the partition walls being each formed to project outwardly between adjacent twos of the vane plates in a radial direction of the impeller and each having two sloped surfaces respectively facing in axial directions of the impeller, a casing which rotatably accommodates the impeller, defines a pump flow passage along the outer periphery of the impeller and has an intake port and a delivery port both communicating with the pump flow passage, and a motor for driving the impeller to rotate the same, characterized in that

said two sloped surfaces (3221, 3222) of each partition wall (321) are formed such that imaginary extensions of said two sloped surfaces intersect with each other at a point (V1) positioned inside a circumferentially facing surface of each vane plate (323) with respect to a radial direction of said impeller (32), each said partition wall (321) has a distal end face (3210, 163), and said distal end face is positioned inside an outer peripheral end (3230) of each said vane plate (323) and joins said two sloped surfaces (3221, 3222) to each other.

12. A fuel pump according to claim 11, characterized in that said distal end face is a flat surface (3210).

13. A fuel pump according to claim 11, characterized in that said distal end face is a curved surface (163).

14. A fuel pump according to claim 11, characterized in that said impeller (32) is formed by molding such that the distal end face (3210, 163) and the sloped surfaces (3221, 3222) of each said partition wall and the circumferentially facing surfaces of each said vane plate (323) remain as they are after the molding, whereas said outer peripheral surface (3230) and axial lateral surfaces (3231, 3232) of each said vane plate are ground after the molding.

15. A fuel pump according to any one of claims 11 to 14, characterized in that the fuel pump is installed in a fuel tank (63) to supply fuel to a fuel injection system (64) of an internal combustion engine (70) and is used with a delivery pressure in the range of 2 to 5 kgf/cm^2 and a delivery rate in the range of 5 to 200 l/h, a diameter (D) of said impeller (32) is in the range of 20 to 65 mm, and a flow passage representative size (Rm) defined by said impeller and said flow passage is in the range of 0.4 to 2.0

mm.

16. A fuel pump according to claim 15, characterized in that a ratio L1/L2 of a distance L1 between said outer peripheral end (3230) of each said vane plate (323) and said distal end of each said partition wall (321) to an entire length L2 of each said vane plate is in the range of 0.1 to 0.6.

17. A fuel pump according to claim 15, characterized in that said two sloped surfaces (3221, 3222) are terminated at their outermost peripheries with a distance in the range of 0.2 to 0.8 mm left therebetween.

18. A fuel pump according to claim 11, characterized in that a surface (3231a) of each said vane plate (323) facing circumferentially is sloped in a rotating direction of said impeller (32).

19. A fuel pump according to claim 11, characterized in that said distal end face of each said partition wall (321) has portions protruding toward said distal end of each said vane plate (323) at joints (3214, 3215) between said distal end face of each said partition wall and adjacent vane plates (323).

FIG. 1

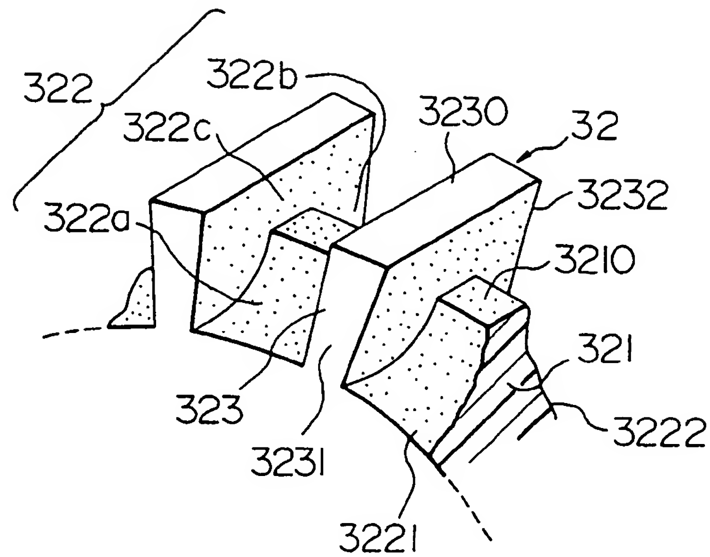

322
322b
322c
3230
32
3232
322a
3210
321
323
3231
3222
3221

FIG. 2

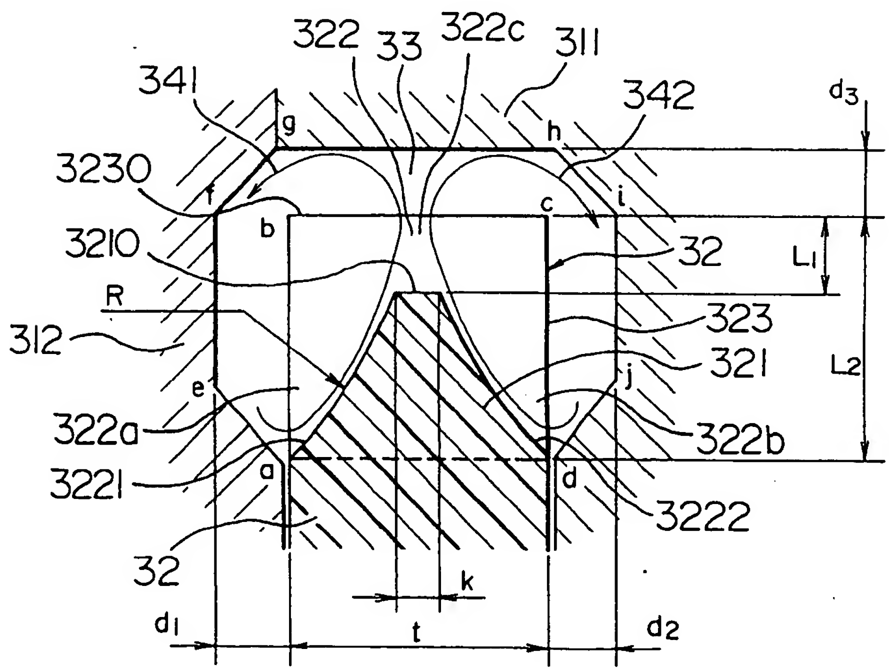

322
33
322c
311
341
342
d3
g
h
3230
f
b
c
i
32
L1
3210
R
323
312
321
L2
e
j
322a
322b
3221
a
d
3222
32
k
d1
t
d2

FIG. 3

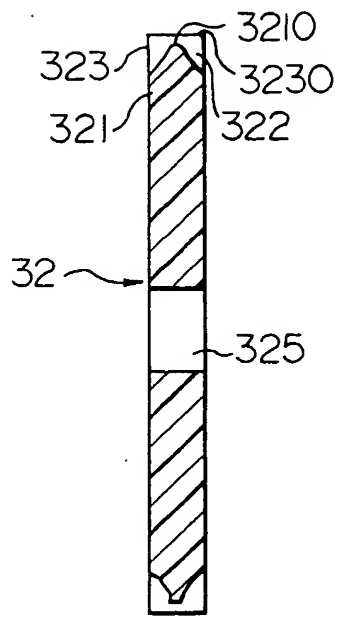

3210
323
3230
321
322
32
325

FIG. 4

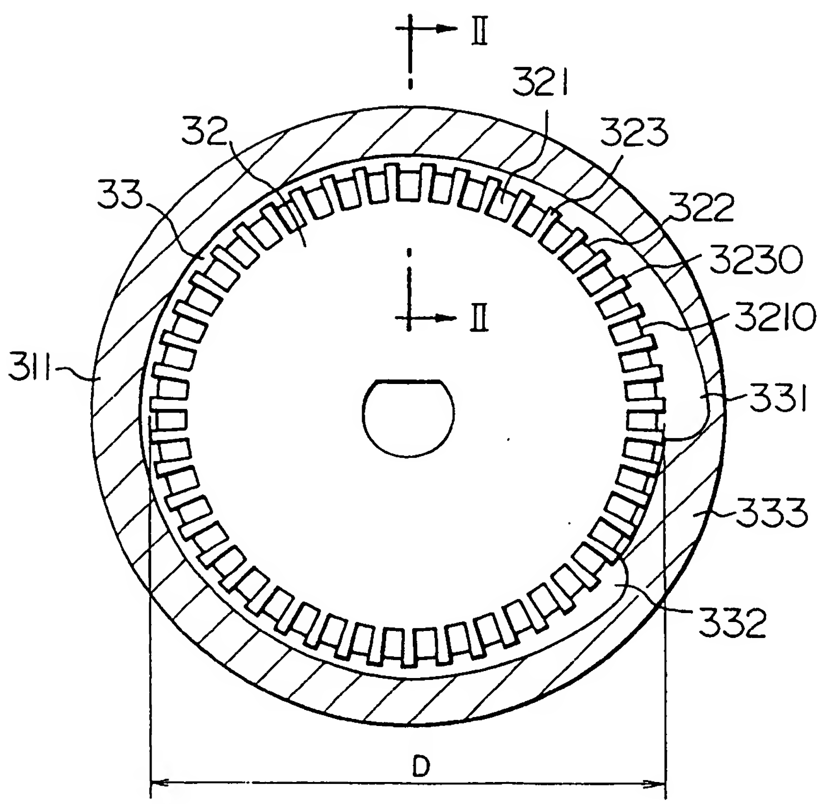

II
321
32
323
33
322
3230
II
3210
311
331
333
332
D

FIG. 5

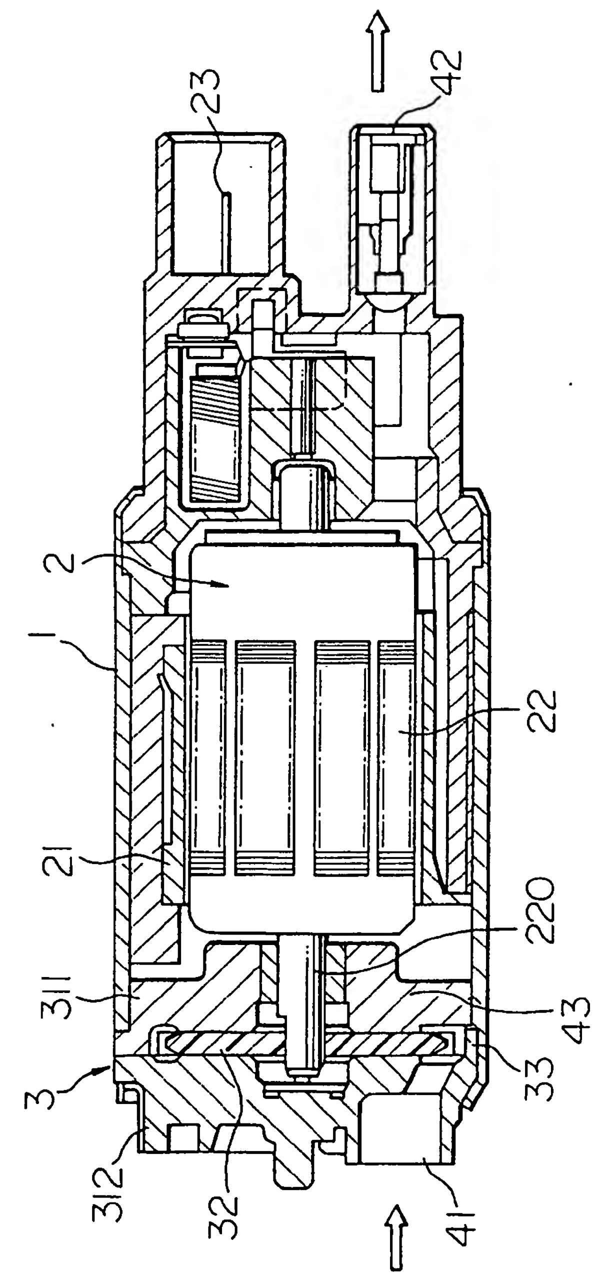


FIG. 6

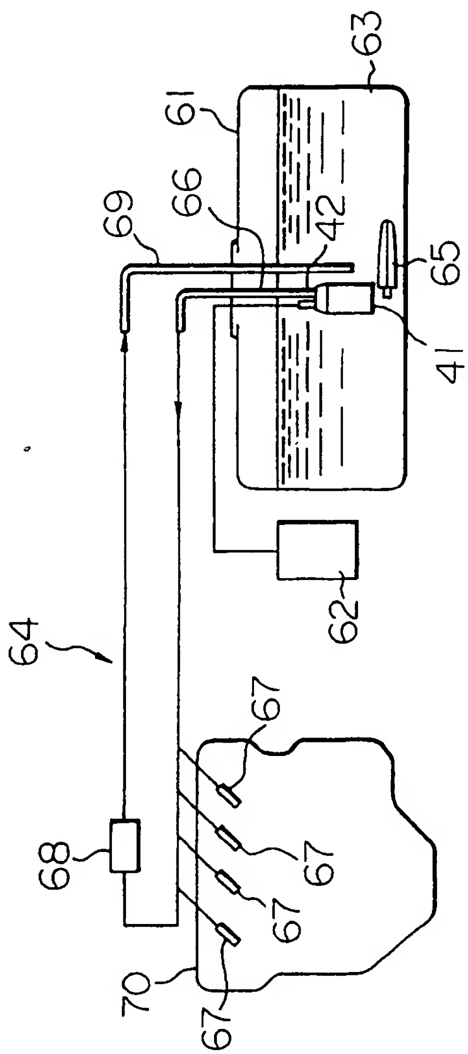


FIG. 7

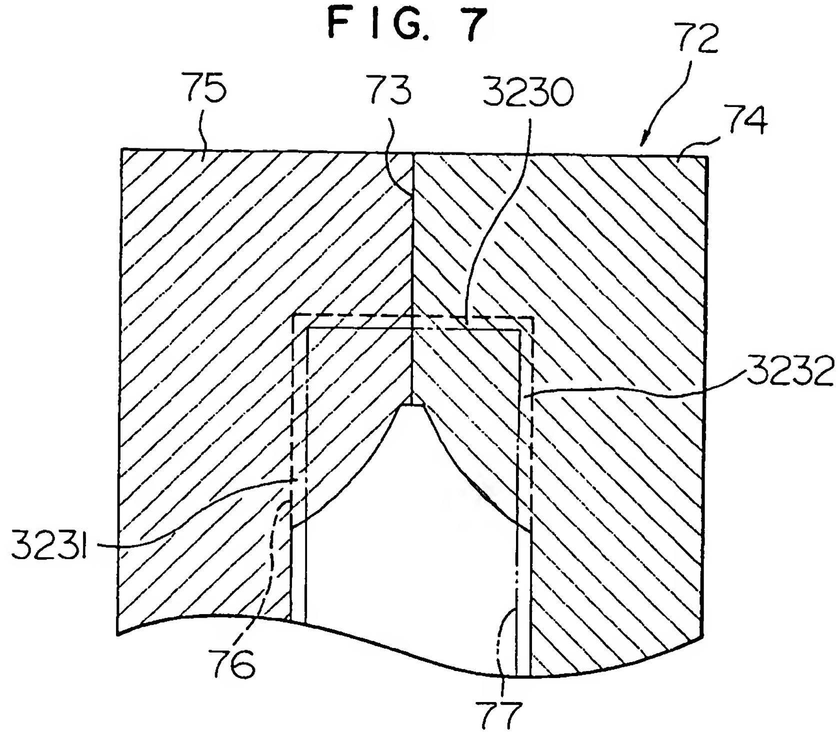

72
75
73
3230
74
3232
3231
76
77

FIG. 8

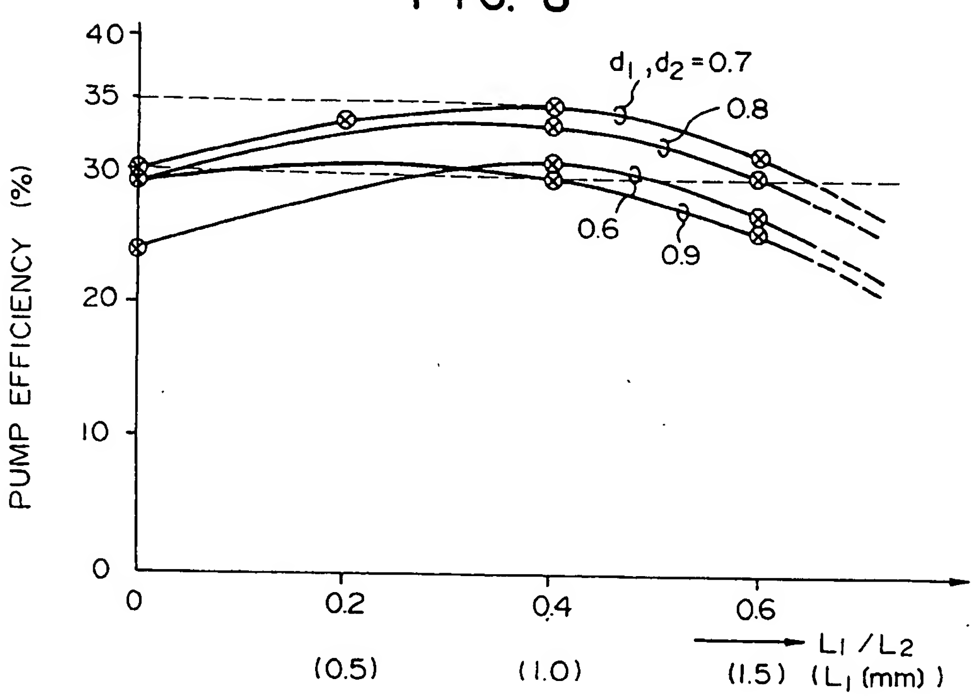

40
35
30
20
10
0
PUMP EFFICIENCY (%)
$d_1, d_2 = 0.7$
0.8
0.6
0.9
0
0.2
0.4
0.6
L_1/L_2
(0.5)
(1.0)
(1.5)
(L_1 (mm))

FIG. 9

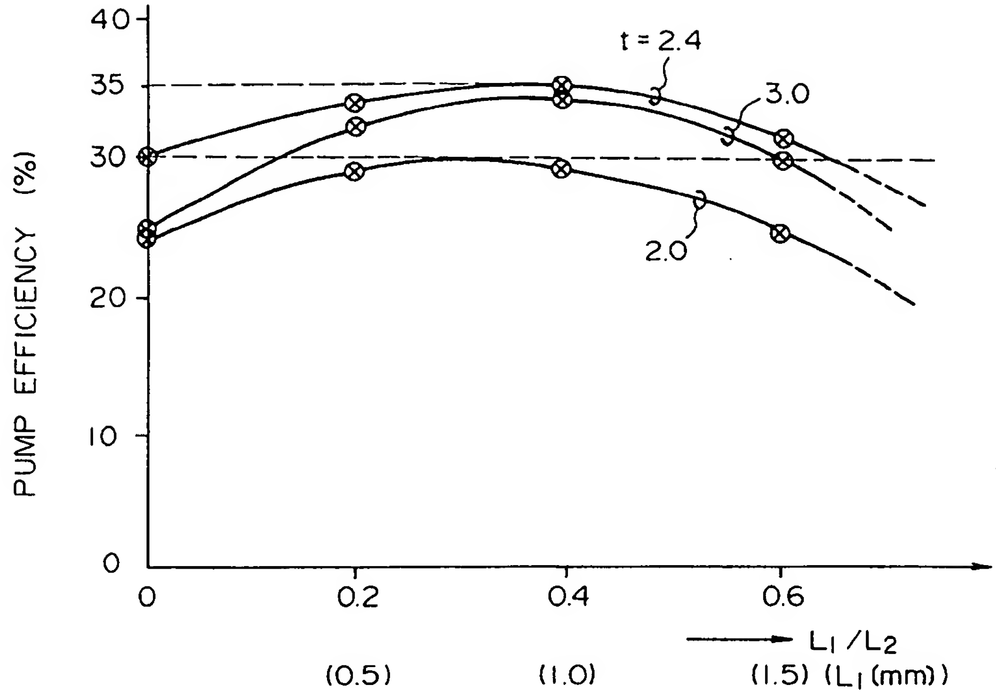

40
35
30
20
10
0
PUMP EFFICIENCY (%)
t = 2.4
3.0
2.0
0
0.2
0.4
0.6
L1 /L2
(0.5)
(1.0)
(1.5) (L1 (mm))

FIG. 10

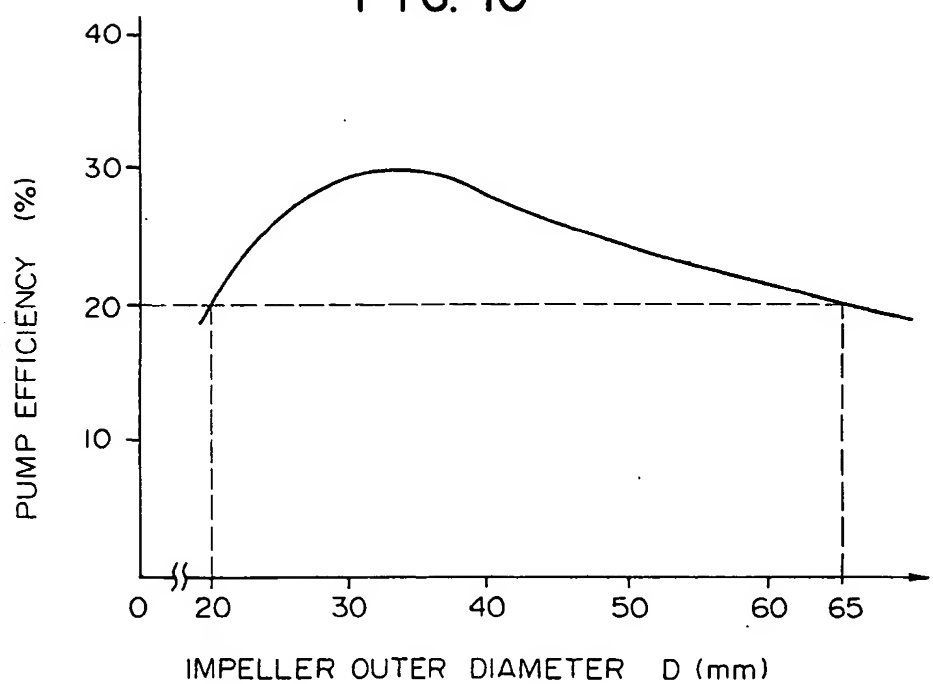

40
30
20
10
PUMP EFFICIENCY (%)
0
20
30
40
50
60
65
IMPELLER OUTER DIAMETER D (mm)

FIG. 11

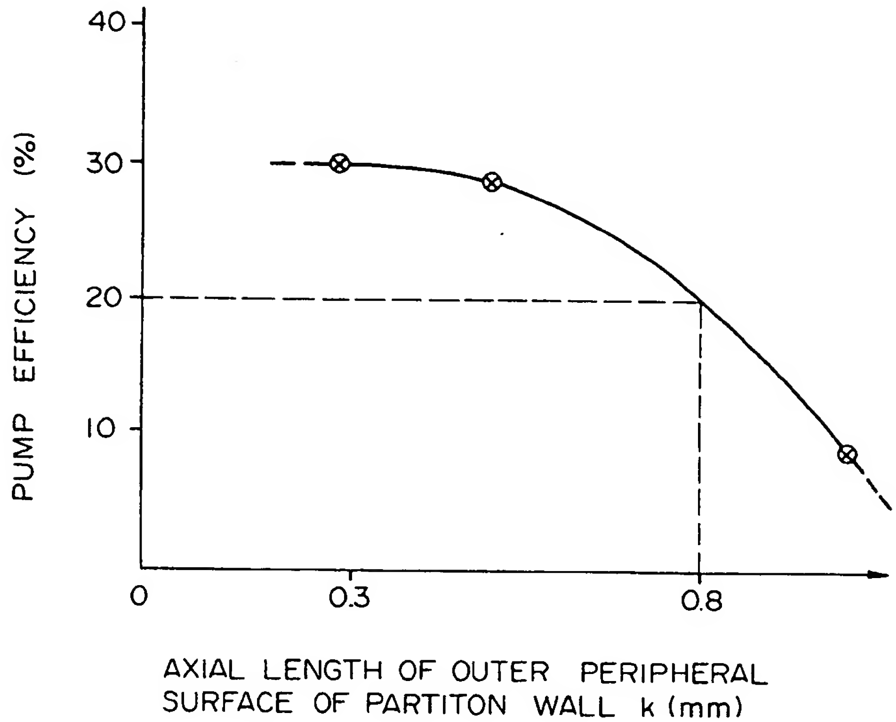

40
30
20
10
PUMP EFFICIENCY (%)
0
0.3
0.8
AXIAL LENGTH OF OUTER PERIPHERAL
SURFACE OF PARTITON WALL k (mm)

FIG. 12

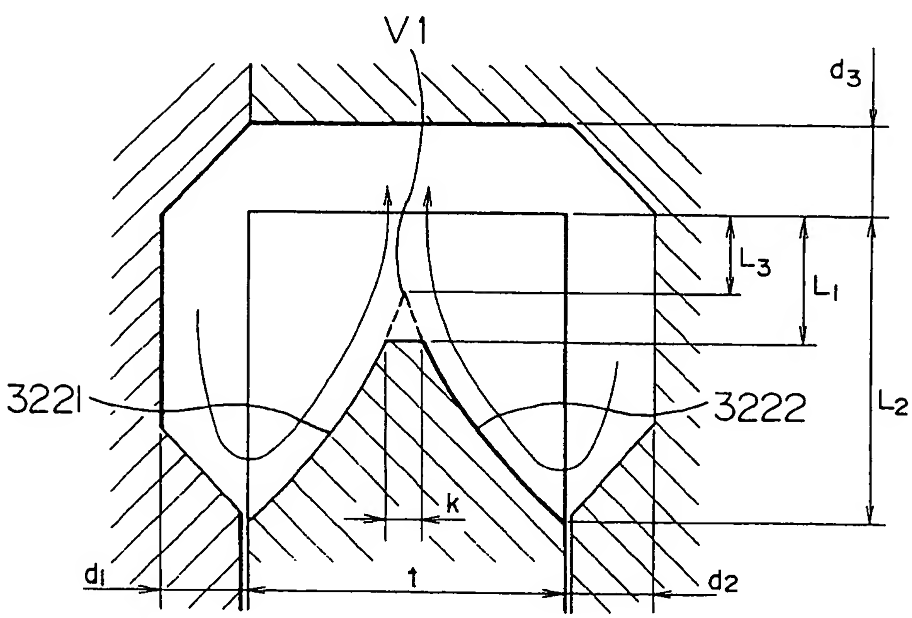

V1
d3
L3
L1
L2
3221
3222
k
d1
t
d2

FIG. 13

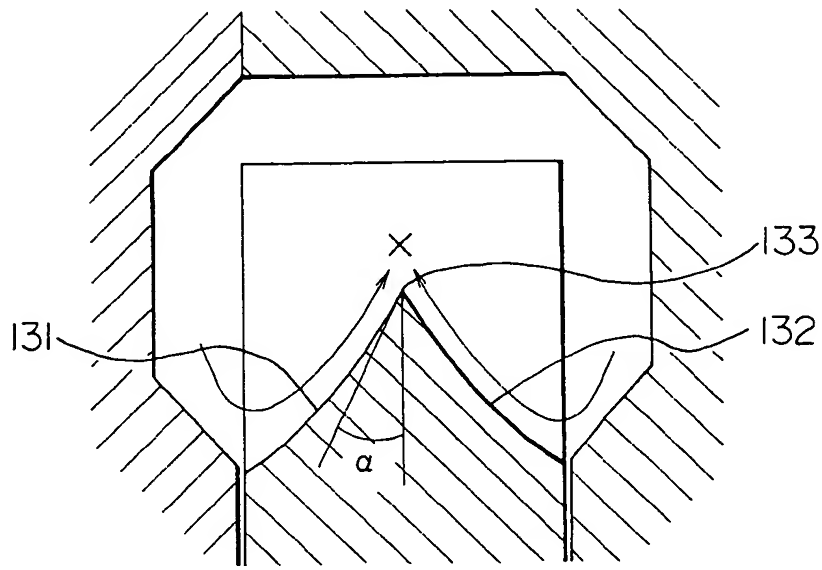


FIG. 14

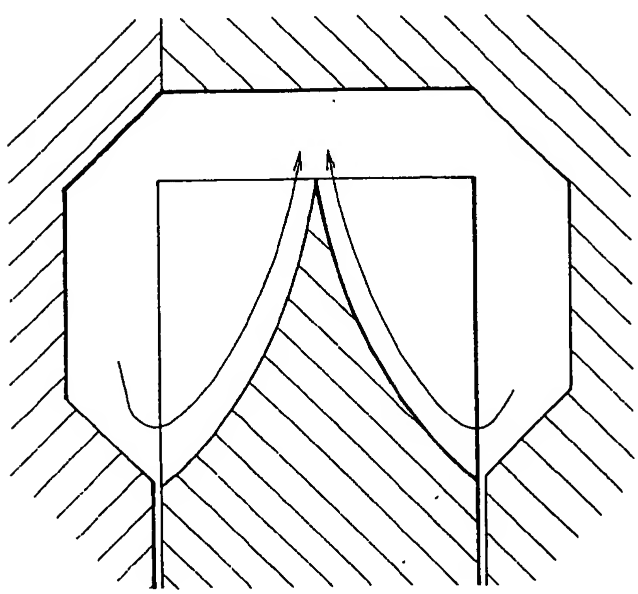


FIG. 15

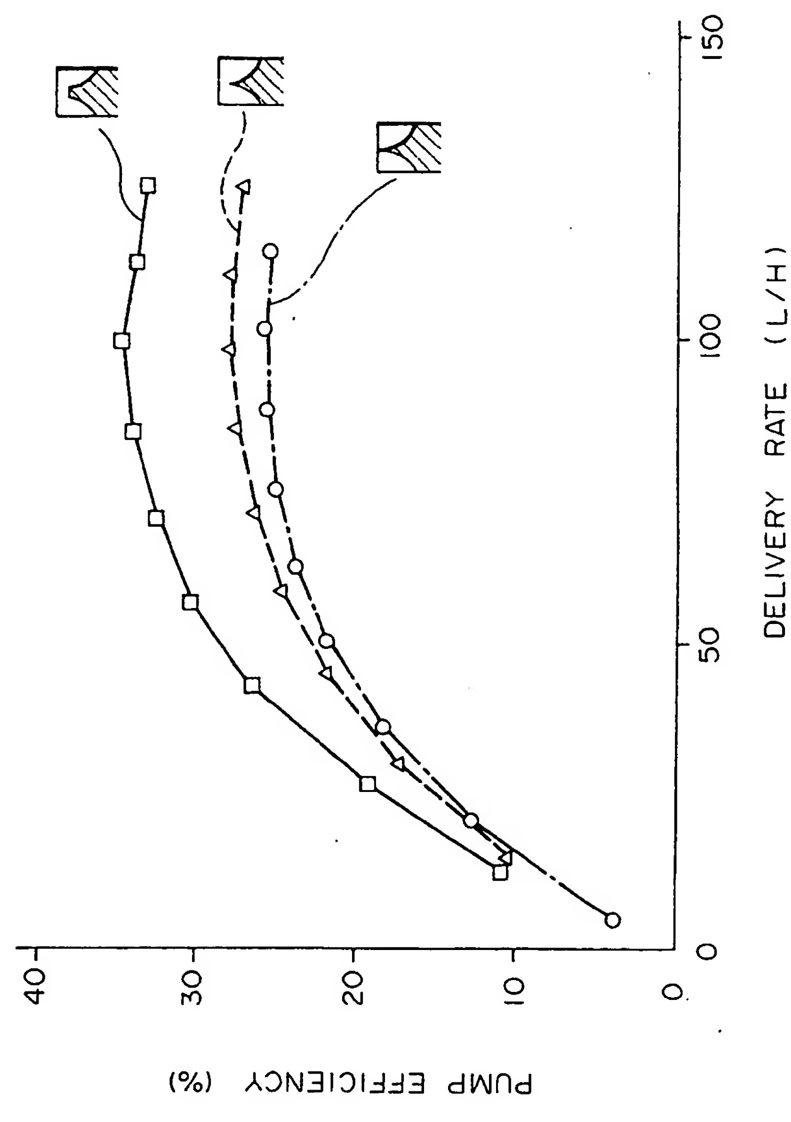


FIG. 16

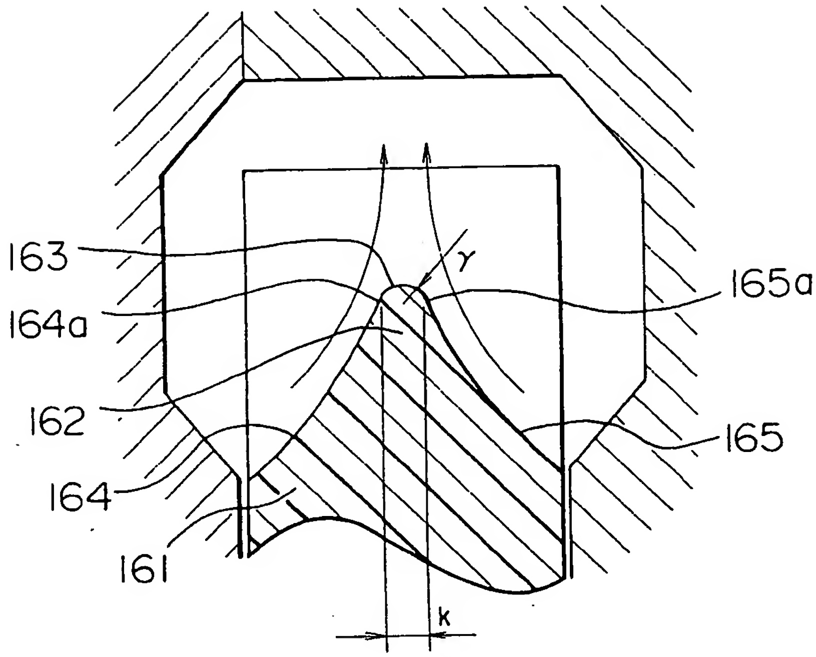

163
164a
162
164
161
165a
165
γ
k

FIG. 17

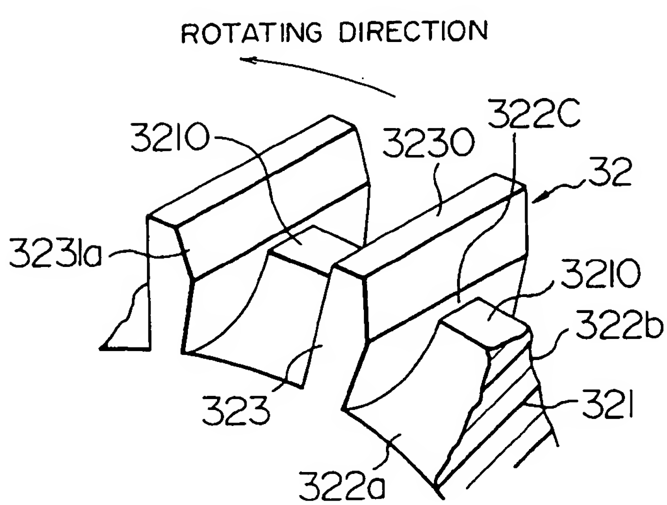

ROTATING DIRECTION
3210
3230
322C
32
3231a
3210
322b
323
321
322a

FIG. 18

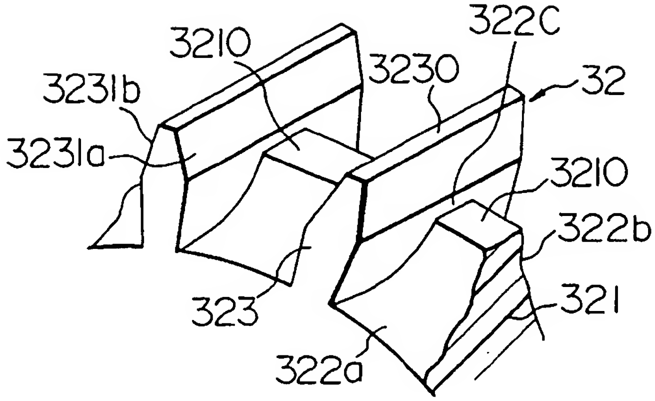

3210
322C
3231b
3230
32
3231a
3210
322b
323
321
322a

FIG. 19

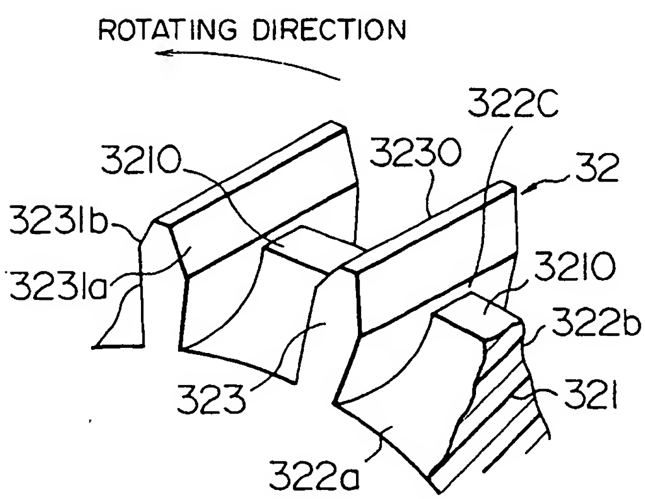

ROTATING DIRECTION
322C
3210
3230
32
3231b
3231a
3210
322b
323
321
322a

FIG. 20

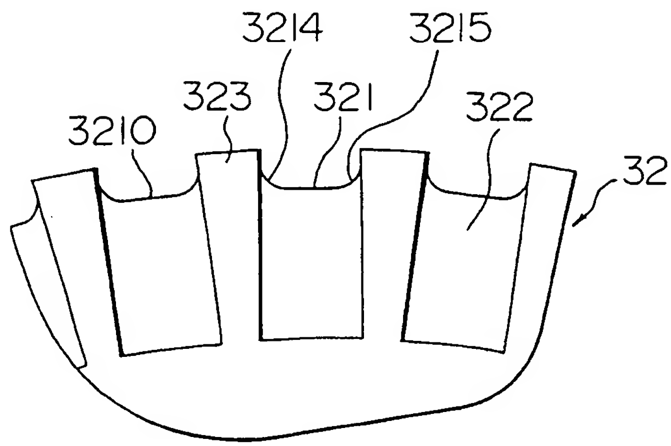


FIG. 21

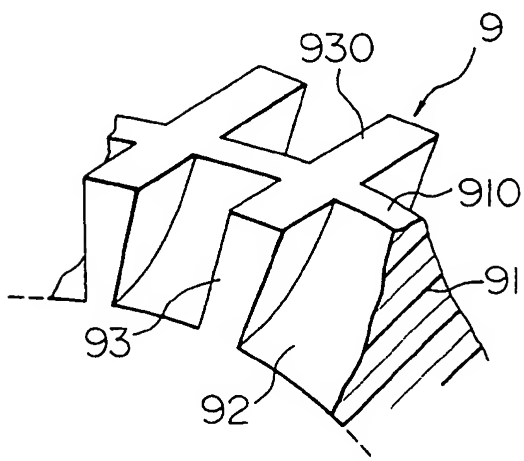
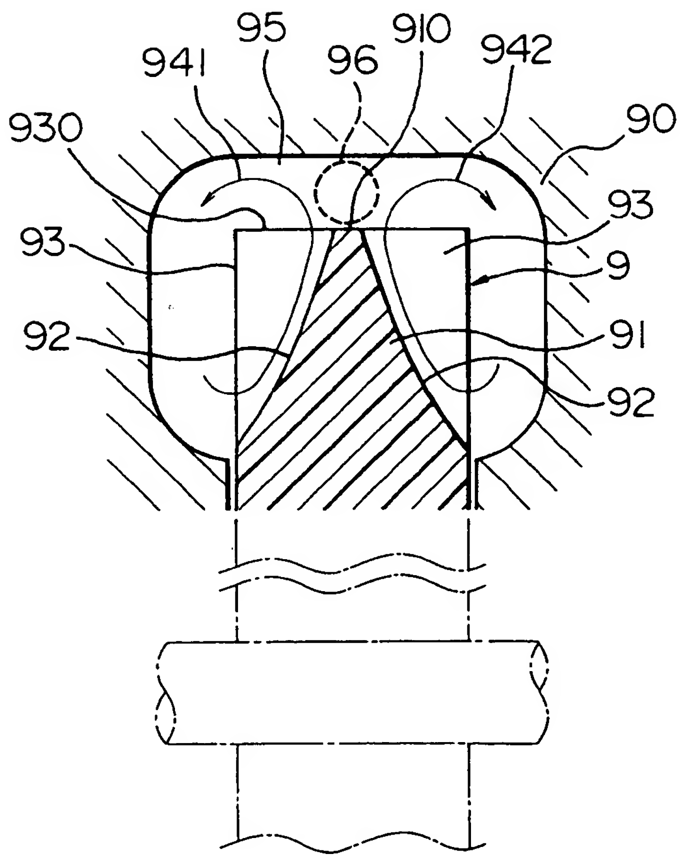


FIG. 22



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EUROPEAN SEARCH REPORT

Application Number

EP 93 10 5414

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.5)
Y	US-A-3 359 908 (TOMA) * abstract * * column 2, line 5 - column 3, line 11; figures * ---	1-3,11, 12	F04D29/18 F04D5/00 F02M37/10
Y	US-A-4 915 582 (NISHIKAWA) * abstract * * column 4, line 19 - column 5, line 41; figures 1-3 * ---	1-3,11, 12	
A	DE-U-8 911 302 (BOSCH) * page 4, line 2 - page 5, line 14; figures * ---	1,2,4,7, 11,13,16	
A	WO-A-9 200 457 (BOSCH) * abstract * * page 2, line 14 - page 4, line 18; figures 1-3 * ---	1,2,4, 11,13	**TECHNICAL FIELDS SEARCHED (Int. Cl.5)**
A	DE-A-3 209 763 (NIPPONDENSO) * abstract; claim 1 * ---	6,15	F04D F02M
A	US-A-4 403 910 (WATANABE ET AL.) * abstract * * column 4, line 63 - column 5, line 13; figures 7A,7B * ---	8,17	
D,A	EP-A-0 133 497 (BOSCH) -----		

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	01 JULY 1993	ZIDI K.

CATEGORY OF CITED DOCUMENTS

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